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Pages including this
cover sheet: 14

To: **Examiner V. Manoharan**

Fax No: **571-273-1450**

From: **Michael Harris**

Re: **U.S. Patent Application No. 10/785,653, Bitterly et al.**

Our file No.: **I022-P07323**

Date: **March 16, 2009**

Remarks: **Attached are the requested specification pages and cover letter.**

PATENT

Matter No. 1022-P07323

Applicants: Bitterly, Jack G. et al.

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Group Art Unit: 1764

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I certify that on **March 16, 2009**,
which is the date I am signing this
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**RESPONSE TO TELEPHONE REQUEST TO SUBMIT
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Sir:

Per your request, we enclose copies of pages 25 through 36 of the specification of this application.

Please either telephone or email me so that I know you received this correspondence and that the attached pages comply with your request.

March 16, 2009

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Boiler and Condenser Surface Area Design: To increase the system water processing output flow, the boiler and condenser surface area must be as large as possible and still be cost effective and practical. The output flow is linearly proportional to the boiler surface area as shown in Equation (6). One unique feature of the water processing system of the present invention is that it uses a common boiler and condenser wall surface. For efficient packaging and provided high heat transfer can be achieved, designing the boilers and condensers to be concentric shells with small separations between them and with decreasing radii inside one large cylindrical container, represents the best space saving layout. The analysis that follows presents relations that compute the total surface area of the combined boiler and condenser surfaces for one embodiment of the present invention. These relations predict the weight and cost of the system and the output flow performance.

The application has discussed three adjacent, concentric shells such as shells **32**, **34** and **40** (FIG. 1). Another way of considering the shells is to think of them as pairs having a boiler and condenser shell. Thus, the system has a set of boiler shells and a set of condenser shells. One of the boiler shells and the adjacent condenser shell form a group pair about a common axis. The next boiler and condenser shell adjacent to the first-mentioned pair, also form a pair about the same axis.

As FIGS. 2 and 3 show, a fixed radial spacing ΔR exists between each shell. In theory, the spacing can vary, but the exemplary embodiment provides fixed spacing. The following relations compute the total area of all boiler and condenser shells:

$$A_{Total} = 2\pi L \sum_{i=1}^N R_i = 2\pi L [R_1 + R_2 + R_3 + \dots + R_N] = 2\pi L \left[R_1 + \sum_{i=2}^N R_i \right]. \quad (11)$$

In this expression N is the number of boiler and condenser shells given by:

$$N = 12 \frac{(R_N - R_1)}{\Delta R} + 1. \quad (11a)$$

This expression has the following definitions:

$$R_1 \text{ \& } R_N = \text{Inner and outer shell radii; (ft)} \quad (11b)$$

$$\Delta R = \text{Separation between each shell; (inches)} \quad (11c)$$

$$R_i = \text{Radius of each shell } i; \text{ (ft)} \quad (11d)$$

$$L = \text{Length of all shells; (ft).} \quad (11e)$$

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Designing the entire system to have a common separation ΔR between each shell minimizes production costs and maximizes output flow capacity. The following relations exist between the shell radius of the i^{th} shell and the shell separation ΔR :

$$\begin{aligned} R_2 &= R_1 + \Delta R / 12 \\ R_3 &= R_2 + \Delta R / 12 = R_1 + 2\Delta R / 12 \\ R_4 &= R_3 + \Delta R / 12 = R_1 + 3\Delta R / 12 \\ R_i &= R_{i-1} + \Delta R / 12 = R_1 + (i-1)\Delta R / 12 ; i = 2, N \end{aligned} \quad (12)$$

5 Therefore, from equations (11) and (12) above, the sum of the radii is given by:

$$\sum_{i=2}^N R_i = \sum_{i=2}^N R_1 + \sum_{i=2}^N (i-1) \left(\frac{\Delta R}{12} \right) = (N-1)R_1 + \sum_{i=2}^N (i-1) \left(\frac{\Delta R}{12} \right). \quad (13)$$

The second sum term in equation (13) is evaluated incorporating the following terms:

$$\sum_{i=2}^N (i-1) \left(\frac{\Delta R}{12} \right) = \left(\frac{\Delta R}{12} \right) [1 + 2 + 3 + \dots + (N-3) + (N-2) + (N-1)]. \quad (14)$$

10 In the summation algorithm, each inner- and outer-most term adds up to N , and there are $(N-1)/2$ of those terms as shown:

$$\begin{array}{c} i + (N-i) \\ \vdots \\ 3 + (N-3) \\ 2 + (N-2) \\ 1 + (N-1) \end{array} \quad (14a)$$

Therefore, the sum given by equation (14) is:

$$\sum_{i=2}^N (i-1) \left(\frac{\Delta R}{12} \right) = N \frac{(N-1)}{2} \left(\frac{\Delta R}{12} \right). \quad (15)$$

15 Substituting equations (13) and (15) into equation (11) gives the total area of all the boiler and condenser shells, which equals:

$$A_{Total} = 2\pi L N \left[R_1 + \frac{(N-1)}{2} \frac{\Delta R}{12} \right] \quad (16)$$

$$A_{Total} = 12\pi L \frac{(R_N + R_1)}{\Delta R} \left[(R_N - R_1) + \frac{\Delta R}{12} \right]. \quad (17)$$

In that equation, A_{Total} = Sum total of all the shell areas (ft²) (17a)

20 R_1 = Inner most shell radius (ft) (17b)

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R_N = Outer most shell radius (ft) (17c)

ΔR = Separation between shells (inches) (17d)

N = Number of boiler shells (17e)

The weight of the boiler and condenser shells can be determined from their known weight density, shell thickness t and surface area A_{Total} . The weight is given by:

$$w_{shell} = \rho_{shell} V = \rho_{shell} t A_{Total} \quad (18)$$

$$w_{shell} = \rho_{shell} t 288\pi LN \left[R_1 + \frac{(N-1)\Delta R}{2} \right] \quad (19)$$

$$w_{shell} = \rho_{shell} t 1728\pi L \frac{(R_N + R_1)}{\Delta R} \left[(R_N - R_1) + \frac{\Delta R}{12} \right] \quad (20)$$

In these expressions, ρ_{shell} = Weight density of boiler/condenser shells (lbs/in³) (20a)

t = Shell thickness (inches). (20b)

Energy Requirements: As discussed previously, the source of energy or heat required for the boiling and condensing process comes entirely from the vapor compressor system 220. The system is adjusted to operate near a boiling water vapor pressure that is commensurate with its ambient temperature. The proposed compressor creates a pressure ratio of about 1.05 to 1.25 on the water vapor that is able to boil at ambient temperature conditions (~70°F or 21°C) about 0.5 psi (0.035 kg/cm²) or less. Consequently, the work done by the compressor is very low. See, e.g., Keenan, J.H. and Keyes, F.G., "Thermodynamic Properties of Steam" John Wiley & Sons, 1936, pp. 28-31.

Though the Keenan and Keyes process envisions a piston-cylinder pressure and volume change, conservation of energy makes the process general and applicable to all compressor designs. Only the input and output thermodynamic states (pressures) and the specific type of thermodynamic process are necessary to predict the energy requirements of the system.

Beginning from first principles, the following discussion presents an analysis of the compressor power requirements for the distillation process. This analysis computes the work performed by the compressor on the fluid by considering the process to be adiabatic, i.e., no heat energy flowing into or out of the compressor other than what is carried by the work done on the compressed fluid. For each "cycle" of the compressor, the relative amount of heat transferred to the compressor and its surroundings compared to the

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amount of heat transferred to the fluid is small and decreases with increased compressor efficiency. Inefficiency losses of the compressor result in reduced laminar kinetic flow of the vapor and increased turbulent kinetic flow. These losses are minimized with the appropriate compressor design. Still, the inefficiency losses of the compressor can still be applied to the end result to determine the approximate overall energy requirements for this distillation process.

In general, the work or energy required to compress a gas in a cylinder by a distance $d\vec{s}$ can be computed by noting that:

$$W = \int dW = \int_{s_1}^{s_2} \vec{F} \cdot d\vec{s} = \int_{s_1}^{s_2} pA ds = \int_{V_1}^{V_2} p dV. \quad (21)$$

In that equation, W = Work or energy of compression (ft-lbs) (21a)

p = Gas pressure (lbs/ft²) (21b)

V = Gas volume (ft³). (21c)

The subscripts 1 and 2 in equation (21) refer to the two different states, i.e., input and output volumes or positions. These volume states convert to pressure states by an adiabatic gas law transformation. In this case, the input pressure p_1 is the boiling vapor pressure p_B while the output pressure p_2 is the condensing vapor pressure p_C . Sears, F. W. and Salinger, G. L., "Thermodynamics, Kinetic Theory, and Statistical Thermodynamics" 3rd Ed., Addison-Wesley 1975, pp. 108-109, described the governing equations. For adiabatic processes, the initial and final thermodynamic states are related by the following constant condition:

$$p_1 V_1^\gamma = p_2 V_2^\gamma = p V^\gamma = K. \quad (22)$$

In this expression K is a constant which can be eliminated, and γ is the ratio of specific heats at constant pressure to constant volume. The units for specific heat are (BTU/(lb °F)). The specific heat ratio γ , is defined by:

$$\gamma = \frac{C_p}{C_v} \quad (23)$$

and the specific heats are related to the gas molecular weight given by:

$$R = 778.6(C_p - C_v) = \frac{1544}{\text{Gas Molecular Weight}} \quad (24)$$

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In this expression R is the universal gas constant with units of (Ft-lbs/(°F lb)). The work done by the compressor on the fluid can be computed by performing the integral given in equation (21). Substituting equation (22) into equation (21) and simplifying yields:

$$W = C_w \frac{\gamma}{(\gamma - 1)} p_1 V_1 \left[\left(\frac{p_2}{p_1} \right)^{(1 - \frac{1}{\gamma})} - 1 \right]. \quad (25)$$

5 The units of work in equation (25) are in Watt-hours with the constant conversion factor C_w , which equals:

$$C_w = \text{Constant conversion to Watt-hours} = (746 / (550 \times 3600)). \quad (25a)$$

10 Computing the required energy per pound of water distilled from the device is useful for specifying the energy efficiency of the thermodynamic compressor process. Typical units are in watt-hours per pound (W hr/lb). This can be computed by first determining the rate of energy expenditure, which is the time derivative of equation (25). During a continuous flow process, the input and output pressures are in a steady state condition. Consequently, the only temporal variable is the input volume V_1 , flow rate. Therefore, the equation yields:

$$15 \quad \dot{W} = C_w \frac{\gamma}{(\gamma - 1)} p_1 \dot{V}_1 \left[\left(\frac{p_2}{p_1} \right)^{(1 - \frac{1}{\gamma})} - 1 \right]. \quad (26)$$

In this expression \dot{W} is the power (watts) required for vapor compression.

Now \dot{V}_1 can be computed from the mass flow and density of the boiling vapor. This connects the energy requirements with the boiler and condenser mass flow output. Examining of units reveals the following relationship:

$$20 \quad \dot{V}_1 = \frac{\dot{w}}{\rho_1} \quad (27)$$

$$\text{Where, } \dot{V}_1 = \text{Boiling vapor volume flow rate (ft}^3/\text{hr)} \quad (27a)$$

$$\dot{w} = \text{Mass flow of fluid vaporized (lbs/hr) given by equation (20)} \quad (27b)$$

$$\rho_1 = \text{Vapor density of the boiled fluid (lbs/ft}^3\text{)}. \quad (27c)$$

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The energy expenditure per pound of water produced E_{ρ}^{100} , for a perfectly efficient system, is computed by $\left(\frac{\dot{W}}{\dot{w}}\right)$. By substituting equation (27) into equation (26), the expected ideal "energy density" becomes:

$$E_{\rho}^{100} = C_w \frac{p_1}{\rho_1} \frac{1}{(\gamma - 1)} \left[\left(\frac{p_2}{p_1} \right)^{\left(1 - \frac{1}{\gamma}\right)} - 1 \right]. \quad (28)$$

Inefficiency losses of the vapor compressor and motor subsystem increase the actual energy density. If their combined inefficiency loss is ξ , the actual energy density is:

$$E_{\rho}^{\xi} = \frac{E_{\rho}^{100}}{\xi}. \quad (29)$$

The ratio of specific heats is required to solve the energy density E_{ρ}^{ξ} . Equation (24) is useful to eliminate from equation (23) the specific heat at constant volume, because only the specific heat at constant pressure is measurable. Substituting equation (24) into equation (23) yields:

$$\gamma = \frac{C_p}{C_p - \frac{1544}{778.6(MW)}}. \quad (30)$$

In this expression MW is the molecular weight (pounds/mole), which is 18 for water or water vapor.

The preceding equations are both temperature and pressure dependent, so the optimum thermodynamic cycle is a complex but determinable relation dependent upon the specific fluid type, ambient temperature and pressure, and contaminants in the fluid.

To appreciate the required energy consumption per pound of water produced by this process, computations were performed using equation (28) for a perfectly efficient vapor compression and motor system. Figure 28 shows a plot of these results.

The computations performed using equation (28) implicitly include the temperature dependent effects on the vapor density, the specific heat of the vapor and the input and output pressure states of the vapor. These effects are not explicitly shown in the equation.

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In the computation process, however, accurate curve fits using fifth order polynomials include these temperature dependencies.

In FIG. 28, the boiler pressure is p_1 or p_B and the condenser pressure is p_2 or p_C . Several families of pressure ratios (p_2 / p_1) are shown for comparison. The displayed pressure ratios are not directly apparent, but successful operation should occur with a boiler and condenser temperature differential of 4°F (2°C) or less. By reading the temperature values from the water vapor boiling curve of FIG. 29, temperature values can be assigned to these pressure values.

In FIG. 29, the increased work required to compress the gas as its density increases with pressure increases causes the general increasing trend for the family of curves of constant pressure ratio. It is not directly apparent that the condenser pressure increases faster than the boiler pressure with increasing boiler pressures. This is true, however. Therefore, the pressure differential Δp between the boiler and condenser must increase for constant pressure ratios and increasing boiler pressures. Consequently, the condenser pressure increases faster than the boiler pressure. Further, a faster increase in condenser pressure translates to an increase in condenser vapor density. This then causes the slight increase in compressor work for increased boiler pressures at fixed pressure ratios.

The family of curves shown in FIG. 29 has a decreasing work dependency as boiler temperatures increase. As the temperature increases for fixed boiler and condenser temperature differentials ΔT , the pressure ratios, (p_2 / p_1), decrease. See FIG. 28. Similarly, the vapor densities decrease with increased temperature. Therefore, slightly less work is required to compress the gas.

As previously mentioned, the motor and compressor inefficiencies increase actual system energy density. See equation (29). Figure 30 shows a sample computation of the system energy density for a wide range of efficiency losses and for several different thermodynamic cycles having varying boiler and condenser differential temperatures ΔT .

The data used in FIG. 30 are the energy density values corresponding to a boiler temperature of 70°F (21°C) with no inefficiency losses. Large electrical motors operate with efficiencies ranging from 93% to 95%. Therefore, 94% is a reasonable motor efficiency ξ_M . Likewise, jet aircraft compressors have efficiencies of 85% to 90% or better.

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Thus, one expects a conservative compressor efficiency ξ_C of 85%. The combined overall system efficiency should be equal to or greater than the following:

$$\xi = \xi_M \times \xi_C = (94\%)(85\%) = 80\% . \quad (31)$$

The data in FIG. 30 then represents the final expected system energy requirements per pound of water distilled assuming that no wasted heat from these inefficiency losses are utilized. They can be used effectively with appropriate heat exchangers to reduce the energy required even more.

The salinity concentrations present in seawater cause an increase in boiling temperature over pure water at the same pressure and temperature conditions. One can compute the increase in boiling point by using the linear relationship developed by Fabuss and Korosi for determining the boiling point rise of various seawater concentrations. See, e.g., Howe, E.D., "Fundamentals of Water Desalination," Marcel Dekker, Inc., 1974, p. 30, for a description of the linear relations between boiling point rise and salinity concentration. FIG. 31 and FIG. 32 show the graphical relations between the boiling point rise versus salinity concentration ratio at various seawater saturation temperatures. As an example, at ambient temperatures of 75°F (24°C), a triply concentrated salinity ratio results in slightly less than a 2°F (1°C) boiling rise increase. This boiling rise increase is the additional energy required when desalinating brine solutions instead of pure water.

The power costs required for the processes of the present invention can be computed by noting that the primary power source is the compressor shaft power. The system also requires an additional small amount of energy for rotational acceleration of the fluid in the boiler/condenser shells. Applicants estimate that additional energy to be approximately 0.2 W hrs/lb.

Electrical power from utilities and solar, diesel and other generators are well suited for powering the proposed water system. Assuming utility electrical power is utilized, the cost per acre-foot of water produced can be computed by the following simple relation:

$$\frac{\$}{\text{Acre Foot}} = (E_p^s) \frac{\text{N-Hrs}}{\text{lb}} (C_e) \frac{\text{¢}}{\text{KW} \cdot \text{Hrs}} \left(\frac{1\$}{100\text{¢}} \right) \left(\frac{1\text{KW}}{1000\text{W}} \right) \left(\frac{8.3454 \text{ lbs water}}{\text{gallon}} \right) \left(\frac{325,851 \text{ gallon}}{\text{Acre Foot}} \right) \quad (32a)$$

or

$$\frac{\$}{\text{Acre Foot}} = (E_p^s) (C_e) (27.194). \quad (32b)$$

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One acre-foot = 1.23×10^6 liters. In this expression $E_p^{\$}$ is the system energy density (FIG. 22) in Watt-hours per pound and C_p is the cost of electrical power that the utility charges in cents per kilowatt-hour.

In other applications such as toxic waste separation at a chemical plant, the costs for water separation draw much higher revenue per weight of input than ordinary water desalination draws. For these special applications, small systems may be more cost-effective than larger ones. Smaller systems have lower capital system costs even though the power input increases.

Operational considerations are a function of the total operational economics. Factors include the users needs in volume flow rate per dollar power costs, user specified practical system size constraints and capital equipment costs.

The following analysis summarizes the theoretically predicted heat transfer performance values resulting from a small ΔT temperature difference between the boiler and condenser that is being transferred to a thin film in a high g rotational field. This application addressed the effects of this high g rotational field on heat transfer. These higher g configurations (up to or greater than 1000 g) provide much higher heat transfer coefficients. Consequently, the overall boiler surface area can be decreased considerably. This makes manufacturing procedures simpler and less expensive. Accordingly, the total number of shells and the overall weight of the system can be reduced. This yields higher heat transfer coefficients. The system achieves these advantages because of the higher rotational g loads on a thin film boiling fluid.

The general relation which describes the boiling heat transfer coefficient in both the pool and nucleate boiling regimes is based on work by Rohsenow (See, for instance, *Handbook of Applied Thermal Design*, Eric C. Guyer, Editor in Chief, McGraw Hill 1989, pp. 1-79). Guyer references the original work of Rohsenow performed in 1952 using correlated experimental data. The relation developed by Rohsenow has the following general form:

$$\frac{(q/A)}{\Delta T_{SAT}} = h_B = \mu_L \Delta h_{LG} (\Delta T_{SAT})^2 \left[\frac{g_N (\rho_L - \rho_G) / g_{grav}}{\sigma} \right]^{\frac{1}{2}} \left[\frac{C_{PL}}{C_{SF} \Delta h_{LG} P_{RL}^S} \right]^3 \quad (33)$$

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where the variables are defined as below with typical ranges for numerical values presented at ambient input temperatures of 70°F:

h_B = Boiling heat transfer coefficient (150 to 15,000 (BTU/(hr ft²°F))

μ_L = Fluid Dynamic Viscosity (2.33 (lb/(hr ft)))

5 Δh_{LG} = Heat of Vaporization Water (1054.3 (BTU/lb))

ΔT_{SAT} = Saturation Temperature Difference (2 to 10 (°F))

$g_N = C_N DN^2 = \frac{1}{2} \left(\frac{\pi}{30} \right)^2 DN^2$ = Rotational Acceleration (64 to 32,200 (ft/sec²)) (34)

D = Shell Diameter (0.5 to 5 (ft))

N = Rotational Speed (50 to 2500 (rpm))

10 ρ_L = Fluid Weight Density (62.3 (lb/ft³))

ρ_G = Vapor Weight Density (1.15×10^{-3} (lb/ft³))

g_{grav} = Weight to Mass Conversion (32.2 (ft/sec²))

σ = Surface Tension at Liquid Vapor Interface (4.97×10^{-3} (lb/ft))

C_{PL} = Specific Heat Fluid (0.998 (BTU/(lb°F))

15 C_{SF} = Material Experimental Constant (0.0058 to 0.013 (No Units))

$P_{RL} = \left(\frac{C_{PL} \mu_L}{k_L} \right)$ = Prandtl Number of Fluid (6.64 (No Units))

S = Exponent Constant (1.0 for Water & 1.7 All Other Fluids (No Units))

Equation (33) represents the expected boiling heat transfer coefficient assuming no forced flow mixing. Forced flow mixing typically increases boiling heat transfer rates by an order of magnitude when the driving ΔT is below 15°F (8°C). See, e.g., McAdams, W.H., *Heat Transmission*, McGraw Hill, 3d Ed., 1954, p. 378. The fluid film migrates along the boiler surface (with a vertical component of gravity depending upon shell orientation). The fluid also has enhanced movement due to increased longitudinal pressures that rotational forces induce. This mixed mode of heat transfer enhances the boiling heat transfer rates even more than equation (33) alone predicts.

Results of Boiling Heat Transfer Calculations: Figures 19-24 show a series of computations of the general trends, which occur with rotational g's on boiling heat transfer rates. These figures are both three-dimensional surface renditions and two-dimensional

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slice plots for two different boiler surface material conditions and three different ambient input temperatures. Some of the figures show the enhanced boiler heat transfer behavior with increased rotational g for a roughly coated Teflon[®] PTFE surface on a stainless steel shell at ambient input temperatures of 70°F, 90°F, and 110°F. This series employs a material characteristic coefficient that was empirically determined to be $C_{SF} = 0.0058$ for a rough Teflon coat on stainless steel. See, Guyer, pp. 1-79. When using rough Teflon coated stainless steel instead of polished stainless steel, a 262% increase in boiling heat transfer occurs. The second series shows similar results for a different boiler surface, namely polished stainless steel only. Here, the characteristic surface coefficient is $C_{SF} =$

5 0.0080. Physically, the performance is reduced for polished stainless steel compared to a rough surface coated Teflon on stainless steel because the roughened Teflon surface provides more locations for nucleate boiling sites. Surface wetting characteristics increase with the Teflon coating.

Observe that as the ambient temperature increases, the boiling heat transfer coefficient increases for the same temperature differences and same rotational g 's. Physically, this occurs because the heat of vaporization reduces with increasing temperature. Therefore, more heat energy is transferred to a boiling fluid for the same ΔT .

The graphs show some advantage for operating the system at increased input ambient temperatures. This requires heating the incoming contaminated liquid, which costs energy. Of course, if the contaminated liquid comes directly from an industrial process, it may be at an elevated temperature. At higher temperatures, the system could employ an additional external set of heat exchangers to recover some of the increased enthalpy carried out with the pure water and waste brine. This approach may be beneficial for certain applications where very small size capital equipment is required and energy costs are less

20 25 important.

Enhancing Boiling Heat Transfer Rates: One method or design enhancement aimed at increasing the boiling heat transfer is to increase the surface roughness of the boiler wall. An example would be to provide minute "grain-like" thin Teflon[®] coatings on the boiling surfaces. This provides a great multitude of nucleate source points on a material, which is normally a non-wetting surface. This enhances the formation of steam bubbles that immediately rises radially out of the film layer from high surface pressure induced g -

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forces. These high g forces provide a higher buoyancy force for the micro steam bubbles in the high g field and create increased micro convective currents. Achieving these higher induced heat transfer coefficients is not possible with stationary systems in a 1 g environment. Several sources in the literature show that the affect of a wettable coating alone in-

5 increases the heat transfer coefficient by up to 300%. See, Kreith, F., *Principles of Heat Transfer*, International Textbook Co., 2d Ed., 1968, pp. 441-445; and Guyer, pp. 1-79.

Using anodized aluminum with a Teflon coating on the boiler is another possibility. First, aluminum reduces the tangential hoop stress in proportion to the ratio of material densities, and the density of aluminum is about 1/3 that of stainless steel. Consequently,

10 the tangential stress decreases from 7600 psi at 850 g's to about 2525 psi for corrosion resistant aluminum. Aluminum also has very high thermal conductivity compared to stainless steel. Aluminum has a thermal conductivity of about 120 (BTU/(hr ft °F)) compared to stainless which has a value of about 10 (BTU/(hr ft °F)). The importance of high thermal conductivity in the wall material becomes greater when the heat transfer coefficients due to

15 boiling and condensing become less of a limiting mechanism.

High vapor velocities (50 to several hundred feet per second) exiting the individual boiler and condenser shells enhance the boiler heat transfer structures. Sweeping contaminated fluid along the boiler wall surface is the mechanism for this enhancement. The sweeping of liquid creates turbulence, which aids in forced convection heat transfer at the

20 vapor fluid interface.

Forced convection through gravity-induced high pressure injection of brine or contaminated liquid also enhances boiler heat transfer conditions. Fluid moving along the boiler wall surface removes stagnant regions and, thus, greatly enhances forced convection heat transfer characteristics. The annular boiler passage area of the system can be

25 modified to achieve favorable vapor velocity with forced convection enhancement along the axial direction. Longitudinal segment dividers placed between both the boiler and condenser shell passage areas at various azimuth locations also may enhance nucleate boiling by convective mechanisms on the boiler surfaces and should enhance drop-wise condensation on the condenser surfaces.

30 *Condensing Heat Transfer Coefficients and Enhancements:* Condensing heat transfer coefficients are usually higher than boiling heat transfer coefficients. Conse-